LBNL 62446

ERNEST ORLANDO LAWRENCE BERKELEY NATIONAL LABORATORY

Energy Implications of Meeting ASHRAE 62.2

Iain S. Walker and Max H. Sherman

Environmental Energy Technologies Division

November 2007

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program, of the U.S. Department of Energy under contract No. DE-AC02-05CH11231.

Disclaimer

Energy Implications of Meeting ASHRAE Standard 62.2

ABSTRACT

The first and only nation-wide standard for residential ventilation in the United States is ASHRAE Standard 62.2-2004. This standard is being considered for adoption by various jurisdictions within the U.S. as well as by various voluntary programs. The adoption of 62.2 would require mechanical ventilation systems to be installed in virtually all new homes, but it allows for a wide variety of design solutions. These solutions, however, may have different energy costs and non-energy benefits. The authors have used a detailed simulation model to evaluate the energy impacts of common and proposed mechanical ventilation approaches for a variety of climates. These results separate the energy needed to ventilate into the energy needed to condition the ventilation air and the energy needed to distribute and/or temper the ventilation air. The results show that exhaust systems are generally the most energy efficient method of meeting the proposed requirements, mostly due to having the least increase in ventilation relative to an unventilated home.

INTRODUCTION

The purpose of ventilation is to provide fresh (or at least outdoor) air for comfort and to ensure healthy indoor air quality by diluting contaminants. Historically, people have ventilated buildings to provide source control for both combustion products and objectionable odors (Sherman (2004). Currently, a wide range of ventilation technologies is available to provide ventilation in dwellings including both mechanical systems and more sustainable technologies. Most of the existing housing stock in the U.S. uses infiltration combined with window opening to provide ventilation. Sometimes this results in over-ventilation with subsequent energy loss or under-ventilation and poor indoor air quality. Recent residential construction methods have created tighter, more energy-saving building envelopes that create a potential for under-ventilation (Sherman and Dickerhoff (1994)), Sherman and Matson (2002)). Infiltration rates in these new homes average a third to a quarter of the rates in existing stock. As a result, new homes often need mechanical ventilation systems to meet current ventilation standards. McWilliams and Sherman (2005) have reviewed such standards and related factors.

Because of the effects it has on health, comfort, and serviceability, indoor air quality in homes is becoming of increasing concern to many people. According to the American Lung Association (ALA 1999) a number of factors within our homes have been increasingly recognized as threats to respiratory health. The Environmental Protection Agency lists poor indoor air quality as one of the largest environmental threat to our country. Asthma is the leading serious chronic illness of children in the U.S. Construction-defect litigation and damage are on the increase in new houses and some of this increase is related to indoor air quality problems such as moisture. Residential ventilation can address many of these indoor air quality problems.

Traditionally residential ventilation was not a major concern because policy makers believed that, between operable windows and envelope leakage, people were getting enough outdoor air. However, recent research (Price and Sherman (2006)) has shown that these days the majority of occupants do not open windows sufficiently from the point of view of satisfying ventilation requirements, such as ASHRAE 62.2. ASHRAE is in the process of publishing an addendum to ASHRAE 62.2 removing the exception allowing operable windows to meet the whole-house mechanical requirements. Therefore in this study we did not include open windows as a way of complying with ASHRAE 62.2. Over the past three decades, houses have become much more energy efficient. At the same time, the types of materials used in furniture, appliances, and building materials in houses have changed. People have also become more environmentally conscious, not only about the resources they were consuming but about the environment in which they live.

ASHRAE Standard 62.2-2004 is the standard for ventilation for acceptable indoor air quality in low rise residential buildings that (together with its companion Standard 62.1 for all other buildings) represents the current standard for setting ventilation rates. The objective of this study is to evaluate the energy impacts of meeting this standard in new homes.

ASHRAE Standard 62.2 has requirements for whole-house ventilation, local exhaust ventilation, source control, and system requirements. The standard assumes that infiltration contributes 2 cfm/100 sq. ft. (0.1 L/s/m^2). In addition to this infiltration, the prescriptive part of the standard requires whole-house mechanical ventilation rate given by Equation 1:

$$\begin{aligned} Q(cfm) &= 0.01A_{floor}\left(ft^2\right) + 7.5(N+1) \\ Q(L/s) &= 0.05A_{floor}\left(m^2\right) + 3.5(N+1) \end{aligned} \tag{1}$$

Where Q is the required ventilation rate, A_{floor} is the house floor area and N is the number of bedrooms. For most houses the ventilation rate requirements of Equation 1 must be met by mechanical ventilation.

Standard 62.2 also requires local mechanical exhaust in kitchens and bathrooms. Kitchens must have the capacity to exhaust at least 100 cfm (50 L/s) through a range hood or provide 5 kitchen air changes per hour. Bathrooms must have the capacity to exhaust 50 cfm (25 L/s) or have 20 cfm (10 L/s) of exhaust continuously.

METHOD

In order to evaluate the energy impacts of the ASHRAE 62.2 requirements, we used a simulation tool to perform minute-by-minute ventilation, heat and moisture calculations. This simulation tool used small, one minute timesteps to allow the dynamic performance of buildings and their Heating, Ventilating and Air Conditioning (HVAC) components to be simulated. The small timesteps were computationally and analytically intensive but they allowed for direct simulation of temporally complex ventilation controls. The simulations were all performed for a full year of 525,600 minutes. The simulations were of typical new homes with and without mechanical ventilation meeting ASHRAE 62.2. We also included houses with leaky envelopes to represent older homes.

The simulation tool has the ability to account for the HVAC system, house and attic air flow, thermal and moisture transport interactions. The following summarizes key aspects of the simulation tool that were relevant for this study. More details of the simulation tool can be found in Walker and Sherman (2006) and a summary of model validation can be found in Walker and Sherman (2007). The simulation tool combined a mass balance for air flows with a thermal model (including the HVAC system) and a moisture transport model.

The air flow model allowed individual (such as passive vents or flues) and distributed leaks (such as those distributed over a wall) to be placed on the building envelope. A rectangular floor plan was assumed and the envelope leaks were separated into the leaks in each of the four walls. Similarly, each face of the building had floor level and eave height leakage. The attic had leakage in the gable ends and two pitched roof surfaces together with eave height soffits, gable vents, ridge vents and vents in the pitched roof surfaces. The airflow mass balance was performed for two zones: the house and the attic. The attic is particularly important in these simulations because the duct system for heating and cooling is located in the attic for many of the simulations and heat transfer and airflows in and out of the ducts are key components of the building load and ventilation airflow network. The flow through each leak was determined by the air flow characteristics of the leak (flow coefficient and pressure exponent) and the pressure across the leak. The pressure across the leak depended on both wind pressures and buoyancy pressures due to indoor-outdoor temperature differences. The wind pressures included local shielding by adjacent buildings based on the wind shadow shelter method of Walker et al. (1996). The houses were assumed to be in a typical urban environment with houses in a row on a street. The mechanical ventilation systems were integrated into the mass balance as constant flow devices. For the HVAC system, when the system blower is off, the duct leaks in the attic are treated the same way as building envelope leakage (in this case the same as ceiling leakage). With the HVAC blower on, the air flow through supply and return grilles was included for the house and air flow through supply and return leaks was included for the attic space. The blower, duct leak and register flows were all treated as fixed air flow rates that depended on the HVAC operating mode: heating, cooling or ventilating. The house and attic interacted via the air flow through the ceiling (and duct leaks with the HVAC system blower off). The mass flows through all the individual flow paths were combined and the mass balance for the house and attic was solved by adjusting the internal pressures of the two zones. Because the equations are non-linear, a highly robust pressure bisection technique was used to determine the attic and house interior pressure shifts. More details of the mass balance model can be found in Walker et al. (2005).

The thermal model used a lumped heat capacity approach with 16 heat transfer nodes including air in the ducts, house, and attic. At each node the rate of change of thermal energy was equated to the sum of the heat fluxes due to radiation, convection and conduction. This resulted in a set of equations that were linear in temperature and were solved simultaneously using Gaussian elimination. The temperatures were then used in the ventilation model so that new mass flow rates could be calculated. This iteration between the thermal and mass flow parts of the simulation was continued until the attic air temperature changed by less than 0.1^{o}C (0.2°F). Usually fewer than five iterations between thermal and ventilation models were required.

The thermal model included:

- overall UA values for the building envelope to determine heat transfer through the envelope of the house. House insulation levels and window performance were based on International Energy Conservation Code (ICC 2005) requirements.
- solar gain through windows that depends on the solar heat gain coefficient and orientation. The solar part of the model used calculations from ASHRAE Handbook of Fundamentals Ch. 31 (ASHRAE 2005) together with measured solar radiation from weather data.
- material thermal properties for the attic envelope including thermal conductivity, thermal mass and surface properties for radiation. The latter was particularly important for attics whose temperatures depend strongly on solar radiation.
- the mass flows derived by the air flow mass balance. This includes air flows through the attic and house envelope together with duct leaks.
- the heat input or removed by the HVAC equipment including latent removal by air conditioning.
- internal gains used values from ASHRAE Handbook of Fundamentals, Ch. 29 (ASHRAE 2005) based on conditioned floor area and number of occupants (611W for the medium sized house in this study)).
- a radiation heat transfer balance in the attic. This included the effects of solar heating and radiative cooling of the roof deck and the impact on duct and ceiling surface temperatures.

The moisture model was based on mass balance between outside, the house, the attic, supply and return ducts and a mass storage term in the house. Moisture removal for the air conditioner was based on system total capacity and sensible heat ratio (SHR). Indoor moisture source strengths were taken from draft ASHRAE Standard 160P (ASHRAE 2006) but modified assuming that the house was unoccupied for 8 hours per day (resulting in two-thirds of the 160P based generation rate). For the moisture storage a mass transport coefficient and total mass storage capacity were used that were determined empirically by comparing predicted humidity variation to measured field data in houses (from Rudd and Henderson (2006) and Building Science Corporation (2006)) to obtain the same diurnal variability and response to air conditioner related dehumidification. Both the transport and storage terms scale with house size (floor area): the total mass capacity for storage was 12.3 lb/ft^2 (60 kg/m^2) of floor area and the mass transport coefficient was 0.0006lb/(sft^2) (0.003kg/(sm^2)). The resulting damping in indoor air moisture variability is close to the empirical formulation in the Environmental Protection Agency's (2001) Indoor Humidity Assessment Tool that uses a capacitance term to allow only 5-10% of moisture flow to inside to go into the air, and assumes the other 90-95% was absorbed by building contents.

Climates Evaluated

Six locations were used that cover the major US climate zones: Houston, Phoenix, Charlotte, Kansas City, Seattle and Minneapolis. TMY2 hourly data files[1] were converted to minute-by-minute timesteps by linear interpolation. The simulations also used location data (altitude and latitude) in solar radiation and air density calculations. The required weather data for the simulations were: direct solar radiation, total horizontal solar radiation, dry bulb temperature, humidity ratio, wind speed, wind direction, and barometric pressure.

Houses Evaluated

[1] From a website maintained by National Renewable Energy Laboratory (NREL)

Three house sizes were simulated to examine the implicit effect of occupant density in the 62.2 requirements: Small (1000 ft^2 (93 m^2) 2 bedroom single story), medium (2000 ft^2 (186 m^2), 2 story, 3 bedrooms) and large (4000 ft^2 (372 m^2) 2-story, 5 bedrooms). The exterior surface area for wall insulation scaled with floor area and number of stories and was set to 1.54 times the floor area for 2-story and 1.22 times floor area for one story. These values were taken from averages of several thousand new Building America homes[2]. Window area was 18% of floor area, with windows equally distributed in walls facing in the four cardinal directions. Slab on grade construction with furnace, cooling coil and ducts in the attic was assumed for Houston, Phoenix, and Charlotte. Minneapolis and Kansas City had basements and for Seattle a crawlspace was used. These different foundations change the envelope load and, in the case of the crawlspace, the wind pressures on the floor level envelope leaks. For the basement houses, half the ducts were assumed to be in the basement and so had only half the exterior area (for conduction and radiation losses) and leakage because the basements were assumed to be conditioned spaces.

The building envelope tightness for the standard house (used for reference) and the mechanically ventilated houses was based on the air leakage database for new construction (Sherman and Matson (2002)) where the typical Normalized Leakage[3] is NL=0.3 (or about 5.8 ACH50 (Air Changes at 50Pa envelope pressure)). These envelope leakage values are summarized in Table 1 together with Effective Leakage Area at 4 Pa (ELA4). An additional house was also simulated to represent an older leaky home. This house had sufficient envelope leakage to meet the requirements of 62.2 without additional mechanical ventilation. The envelope leakage for the leaky home was calculated using the weather factors from ASHRAE Standard 136 and the airflow requirements from section 4.1.3 of ASHRAE Standard 62.2, i.e.:

$$\text{ASHRAE Standard 136 airflow} = 2\times Q + 2\ \text{cfm}/100\text{ft}^2\ \text{floor area} \qquad \textbf{(2)}$$
$$\text{ASHRAE Standard 136 airflow} = 2\times Q + 10\ \text{L/s}/100\text{m}^2\ \text{floor area}$$

where Q is the whole house ventilation rate from Equation 1.

Table 2 summarizes the weather factors from Standard 136, and the corresponding envelope leakage used in the simulations. The envelope leakage of the leaky house is approximately doubled for the two story houses and tripled for the single story small house relative to the standard house, due to the difference in stack height driving natural infiltration.

For the three house sizes used in this study the ventilation requirements from Equation 1 are:

1000 ft^2 (93 m^2) & 2 bedrooms (3 occupants) $\Rightarrow$ 32.5 cfm (15 L/s)

2000 ft^2 (186 m^2) & 3 bedrooms (4 occupants) $\Rightarrow$ 50 cfm (23 L/s)

4000 ft^2 (372 m^2) & 5 bedrooms (6 occupants) $\Rightarrow$ 85 cfm (40 L/s)

To meet the kitchen and bathroom requirements of ASHRAE 62.2, the simulations included bathroom fans that operated for half an hour every morning from 7:30 a.m. to 8:00 a.m. For houses with multiple bathrooms the bathroom fans operated at the same time, so the medium house had a total of 100 cfm (50 L/s) and the large house had a total of 150 cfm (75 L/s). Power requirements for these fans were 0.9 cfm/W based on recent California field survey data, i.e., 55W for each 50 cfm fan. Note that this is significantly more power than used by the high efficiency ventilation fans. Similarly, all simulations had kitchen fan operation. The kitchen fans operated for one hour per day from 5 p.m. to 6 p.m. Very few kitchen fans have rated air flows as low as the 62.2 air flow requirement. We chose the smallest fan with a power consumption rating listed in the HVI (HVI 2005) directory which had a flow rate of 160 cfm (80 L/s), and used 99 W.

Table 1 Envelope Leakage for Base Case and Mechanically Ventilated Homes

House Size	ACH 50	ELA4 (in^2)	ELA4 (cm2)	m^3/sPa^n	cfm/Pa^n
Small	5.8	43	277	0.028	61
Medium	5.8	86	554	0.057	121
Large	5.8	173	1116	0.114	243

Heating and Cooling Equipment

2 Information provided by Armin Rudd of Building Science Corporation by personal communication

3 Effective leakage area normalized by floor area and house height

Equipment sizing was based on ACCA Manuals J and S (ACCA 1986). Equipment sizing was important in this study because several ventilation systems used the furnace blower to distribute ventilation air. The equipment capacity determined the blower size and power consumption, and contributed to the energy used by the ventilation systems using the furnace blower to distribute ventilation air.

The heating was supplied by a standard 78% AFUE gas furnace. The furnace blower motor waste heat (85% of blower power consumption) was included as an extra heat source. The furnace blower ran for one additional minute with the burners off at the end of each heating cycle to purge the system.

For cooling, a standard Energy Efficiency Ratio (EER) 11 (Seasonal Energy Efficiency Ratio (SEER) 13) split system air conditioner was used with correct air handler flow and refrigerant charge. The air conditioner capacity calculations depended on air flow across the cooling coil (using the methods in ASHRAE Standard 152 (ASHRAE 2004) fixed at a nominal 400 cfm/ton (47 L/s/W)), the outdoor dry bulb temperature and duct return dry bulb temperature and humidity ratio. The duct return dry bulb temperature was changed from indoor conditions by heat transfer to or from the return ducts in the attic. The return air temperature and humidity ratio included the air flow into the return for central fan integrated ventilation systems and return duct leakage. The sensible heat ratio depended on the return air humidity ratio using a simple linear correlation based on manufacturer's data. The latent capacity at the beginning of each air conditioning cycle ramped up over a period of three minutes based on the work of Henderson and Rengarahan (1996) and Henderson (1998). The quantity of moisture condensed on the cooling coil was tracked to account for transient latent performance, short cycling effects and evaporation during fan-only (no cooling) operation used by some of the ventilation systems.

Table 2 Envelope Leakage of Leaky Houses (Required to meet ASHRAE Standard 62.2 Infiltration Requirements without Continuous Mechanical Ventilation)

Location	Weather Factor From ASHRAE 136	Building Envelope Airflow Requirement From Equation 2	Normalized Leakage	Envelope Leakage Coefficient	ACH50
		ACH		m^3/sPa^n	
1000 ft² (93 m²), one story					
Seattle	0.85	0.64	0.75	0.073	13.9
Phoenix	0.68	0.64	0.94	0.091	17.4
Minneapolis	0.97	0.64	0.66	0.064	12.2
Kansas City	0.85	0.64	0.75	0.073	13.9
Charlotte	0.74	0.64	0.86	0.084	16.0
Houston	0.81	0.64	0.79	0.077	14.6
2000 ft² (186 m²), 2 story					
Seattle	0.85	0.53	0.62	0.098	9.3
Phoenix	0.68	0.53	0.77	0.122	11.6
Minneapolis	0.97	0.53	0.54	0.086	8.1
Kansas City	0.85	0.53	0.62	0.098	9.3
Charlotte	0.74	0.53	0.71	0.112	10.7
Houston	0.81	0.53	0.65	0.103	9.8
4000 ft² (372 m²), 2 story					
Seattle	0.85	0.47	0.55	0.175	8.3
Phoenix	0.68	0.47	0.69	0.219	10.4
Minneapolis	0.97	0.47	0.48	0.153	7.3
Kansas City	0.85	0.47	0.55	0.175	8.3
Charlotte	0.74	0.47	0.64	0.201	9.6
Houston	0.81	0.47	0.58	0.184	8.7

The duct leakage was 3% supply and 2% return (expressed as a fraction of the total blower air flow). For Houston, Phoenix, Seattle and Charlotte, these leaks were to and from the attic. For the basement houses in Kansas City and Minneapolis, the leakage was reduced to 1.5% supply and 1% return, because it was assumed that the basements were inside conditioned space and half the ducts were in the basement. This level of leakage is significantly lower than typical new construction (that is about 34% total (supply + return) from the summary of several studies in Walker (1998)), however we decided to use the lower values in this study because sealed ducts are a requirement in the IECC and should be a requirement for ducts used for ventilation.

Operation of the heating and cooling equipment used set-up and set-back thermostat settings. For heating, the setpoint was 70°F (21°C) from 8 a.m. to 11 p.m. and 68°F (20°F) the rest of the time. For cooling, the setpoint was 78°F (25.5°C) from 8 a.m. to 4 p.m. and 76°F (24.5°C) the rest of the time. On weekends the cooling setpoint was not set up to 78°F (25.5 °C) and remained at 76°F (24.5 °C). The deadband for the thermostat was 1°F (0.5°C). Heating and cooling were available every minute of the year and the operation of heating and cooling equipment was solely decided by the indoor temperature compared to the thermostat settings.

Systems Evaluated

We considered nine different ventilation systems derived from the review of Russell et al. (2005). Power requirements for fans giving the required flows were determined from the Home Ventilating Institute directory (HVI (2005)). The ventilation fans were also selected to meet the sound requirements of ASHRAE 62.2 using the listed values in the HVI directory. The following ventilation systems were simulated:

Standard House (no whole-house mechanical ventilation)

This was the base case for comparison to the other ventilation methods and was simulated for all six climates and three house sizes. This was the same house as the mechanically ventilated cases, except it had no whole-house mechanical ventilation, only bathroom and kitchen source control exhaust. This house was not 62.2 compliant.

House with Leaky Envelope that meets 62.2 (no whole-house mechanical ventilation)

This represented existing homes and was simulated for all six climates and three house sizes. This house was 62.2 compliant because of its leaky envelope. Table 2 shows the envelope leakage used for this case.

Continuous exhaust

Continuous exhaust was simulated using a bathroom exhaust fan. For the small, medium and large houses the fan power requirements were 13.1W, 18.1 W and 20.5W.

Intermittent Exhaust: Intermittent exhaust was simulated using bathroom fans that were on for 20 hours and off for 4 hours during peak space conditioning load (3-7 p.m. for cooling and 1 – 5:00 a.m. for heating). The fan flow was increased from the continuous exhaust case to account for the intermittent operation using the ASHRAE Standard 62.2 ventilation effectiveness calculations. For the medium house, the correct flow to obtain the average ASHRAE Standard 62.2 value was 60 cfm (28 L/s), and a fan providing this air flow used 24.3 W.

Heat Recovery Ventilator (HRV) & Energy recovery Ventilator (ERV)

An HRV was used in Minneapolis, Seattle, Kansas City and Phoenix. Houston and Charlotte used ERVs. The HRV and ERVs were connected to the forced air duct system and the furnace blower operated at the same time as HRV and ERV operation to distribute the air. The lowest airflow HRV in the HVI directory had an airflow of 138 cfm (65 L/s) (137 cfm (64.5 L/s) for the ERV) so this airflow was used in the simulations. These airflows were greater than the ASHRAE 62.2 required flow rate for the medium house, therefore the HRV and ERV were operated for 21 minutes out of each hour so that the hourly average rate met ASHRAE 62.2 The HRV used in this study used 124W and had an Apparent Sensible Effectiveness (ASE) of 70% (that was used to calculate the temperature of air supplied by the HRV). The ERV used 126 W had an ASE of 68%, Total Recovery Efficiency of 45% and Latent Recovery of 36% when cooling. The latent recovery fraction was used to calculate the humidity of the air supplied to the house by the ERV. Because the HRV/ERVs used the forced air heating and cooling ducts, there was also an increase in ventilation due to duct leakage. The supply duct leaks range from 25 to 50 cfm (12 to 25 L/s) and return duct leaks range from 17 to 35 cfm (9 to 18 L/s). These are a substantial fraction of the ASHRAE Standard 62.2 required air flow rates.

Continuous Exhaust plus Air Distribution: The continuous exhaust system was augmented with a central fan integrated (CFI) supply that used the furnace blower to intentionally draw outdoor air through a duct from outside into the return and distribute it throughout the house using the heating/cooling supply ducts. The outdoor air duct was only open to outdoors during heating and cooling blower operation and had a damper that closed when the blower was off. The flow through the outdoor air duct into the return was the same as the continuous exhaust fan flow. The damper was assumed to have zero leakage when closed. The air leakage and thermal losses from the HVAC system ducts were included in the operation of this ventilation system.

Continuous Supply: The continuous supply system used a fan to supply filtered air from outside to the duct system that then distributed the air throughout the house without using the furnace blower. A mixing ratio of 3:1 for indoor to supply air was used to temper the air. The supply fan was therefore sized to be four times the ASHRAE Standard 62.2 outdoor air requirements, i.e., 200 cfm (94 L/s) for the medium sized house. The fan rated closest to this flow in the HVI directory provided 205 cfm (96 L/s) and used 79W. Like the furnace blower the waste heat (69W) from this fan was added to the supplied air. Because this supply fan will normally be an inline fan located outside the building thermal envelope an exception in ASHRAE Standard 62.2 means that it does not have to meet the low sone requirement. This is fortunate, as the inline fans in the HVI directory either do not have sone ratings or do not meet the low sone requirements in ASHRAE Standard 62.2.

RESULTS

To compare the different systems, the minute-by-minute results were summed or averaged over the year depending on the particular parameter of interest. The simulation tool calculated minute-by-minute ventilation rates. For an indoor air quality assessment, the average indoor pollutant concentration is required rather than the average air flow rates. To compare the different ventilation systems an effective air change rate was calculated from the simulation air flows. The effective air change rate gives the same dose to an occupant (for a constant generation indoor pollution source) as the variable ventilation rates resulting from the simulations. The relationships developed by Sherman and Wilson (1986) and Yuill (1986, 1991) were used to convert the minute-by-minute air flow data from the simulations to the effective air change rate. The effective ventilation rate is always lower that the mean ventilation rate and the two are exactly equal when ventilation rates are constant. The biggest differences between average air change rate and effective air change rate (shown in Table 3) are for the two non-mechanically ventilated cases because they have the greatest air flow variability during the year.

Table 3 summarizes the effective annual air change rates for the different ventilation systems in the six climates. The climate trends are the same for all the cases. As the climate goes from mild in Houston to the extreme cold of Minneapolis, the average air change rate increases by about 0.1 ACH. The predicted ACH of a leaky house is very sensitive to climate, despite using ASHRAE Standard 136 to estimate changes in envelope leakage for each climate.

The lowest effective ventilation rate was for the standard unventilated[4] house that is not Standard 62.2 compliant.

Of the simulations that included mechanical ventilation systems, the buildings with the HRV/ERV systems stand out as having substantially greater air change rates. This is because these are balanced (i.e., supply and exhaust air flows are the same) ventilation systems that effectively add directly to the natural infiltration. The other systems are not balanced and combine non-linearly in a sub-additive manner with the natural infiltration. The net effect for unbalanced systems is that the additive effect of the mechanical ventilation is reduced (see Wilson and Walker (1990), Palmiter and Bond (1991) and ASHRAE Fundamentals, Ch. 27 (2005)). In addition, the magnitude of the effect of unbalanced supply and exhaust systems are different. This is because the effect of typical wind and temperature induced building envelope pressure differences is to create a lower pressure inside than outside. This means that the sub-additive effects of imbalanced leakage are typically greater for exhaust than supply systems.

The effect of duct leakage and balanced instead of imbalanced operation can be seen by comparing the continuous exhaust with air distribution (that includes supply air to balance the exhaust system) to just the continuous exhaust: the balanced air distribution operation added about 0.02 ACH on average.

[4] In this paper all other houses are considered to be ventilated because we deliberately added envelope leakage or 62.2 complaint mechanical ventilation.

Table 3 Effective Ventilation Rate [Average Ventilation Rate] (ACH)						
City	**Houston**	**Phoenix**	**Charlotte**	**Kansas City**	**Seattle**	**Minneapolis**
Standard Unventilated House	0.16 [0.18]	0.16 [0.18]	0.17 [0.20]	0.20 [0.24]	0.23 [0.24]	0.24 [0.29]
Leaky House	0.24 [0.28]	0.28 [0.31]	0.28 [0.34]	0.32 [0.39]	0.35 [0.38]	0.34 [0.42]
Continuous Exhaust	0.27 [0.28]	0.27 [0.29]	0.27 [0.28]	0.30 [0.32]	0.28 [0.30]	0.32 [0.35]
Intermittent Exhaust	0.27 [0.28]	0.27 [0.29]	0.27 [0.29]	0.30 [0.32]	0.29 [0.30]	0.32 [0.36]
HRV/ERV	0.36 [0.38]	0.38 [0.39]	0.38 [0.39]	0.41 [0.43]	0.42 [0.43]	0.44 [0.47]
Continuous Exhaust with Air Distribution	0.28 [0.30]	0.28 [0.30]	0.29 [0.31]	0.32 [0.35]	0.32 [0.34]	0.34 [0.38]
Continuous Supply	0.30 [0.31]	0.29 [0.29]	0.29 [0.31]	0.35 [0.38]	0.36 [0.38]	0.39 [0.43]

ENERGY USE

Because the details of design features like the building envelope are common, we have elected to show the results relative to a base case, rather than as total. The advantage of this approach is that the common factors are subtracted out and the results reflect the difference due to the ventilation treatment chosen. We have elected to use the *unventilated house* as the base case, but it is important to remember that that case is a non-ASHRAE 62.2 compliant one.

Figures 1-6 show stacked bar charts where we have parsed the total energy use into the energy to induce the desired ventilation (Ventilation), the energy to distribute air (Distribution) and the energy to condition the air (Space Conditioning that combines heating and cooling). The energy use data was converted to dollars ($) assuming gas at $1.55/Therm (equivalent to $0.06/kWh) and the following electric utility rates[5] : Houston $0.115/kWh, Phoenix $0.095/kWh, Charlotte $0.093/kWh, Kansas City $0.085/kWh, Seattle $0.069/kWh and Minneapolis $0.06/kWh.

Figure 1 shows the results for Houston where the total energy use for the standard house was 16,500 kWh (cost was $1125). The exhaust systems had the least energy increase of about 800 to 1000 kWh (5% of the total space conditioning) and the intermittent system saved 200 kWh ($18) relative to the continuous system. Both the ERV and Continuous Supply systems used significant distribution energy that made them the least energy efficient. The higher ventilation rates of the continuous supply and continuous exhaust plus intermittent supply lead to greater space conditioning energy use. Although the ERV had the greatest air change rates, the recovery of energy was shown by having the least space conditioning energy use. However the distribution energy requirements and ventilation fan power made the ERV the greatest energy user (costing an additional $240 compared to the standard house). In comparison to the leaky house, both the exhaust only systems actually used less energy despite providing greater effective ventilation. The leaky house suffers from having higher ventilation rates when indoor outdoor temperatures are greatest leading to excess energy use.

---

[5] Energy Information Administration, Form EIA-826, "Monthly Electric Sales and Revenue Report with State Distributions Report."

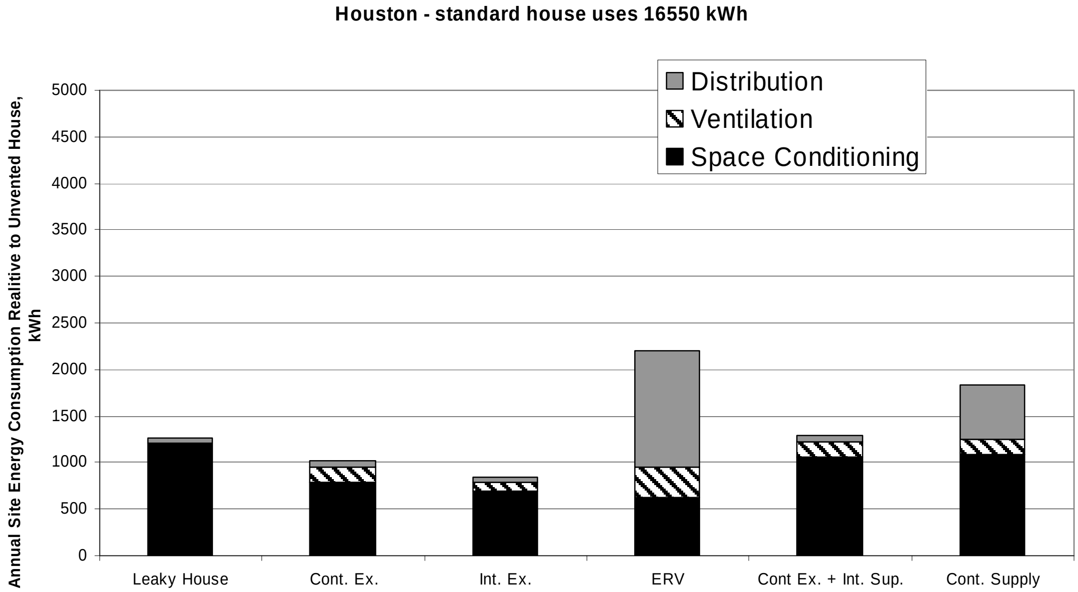

Figure 1. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Houston.

Figure 2 shows the results for Phoenix where the total energy use for the standard house was 16,600 kWh (cost was $1170). Although this is considered a cooling-dominated climate (with almost four times as much cooling operating time than heating), the energy used for heating (9600 kWh) is still greater than that used for cooling (5500 kWh). The Phoenix results had the same general trends as Houston, but the milder climate in Phoenix means that the space conditioning energy use related to ventilation is lowered. The exhaust system used 800 kWh ($60). The intermittent exhaust saved 160 kWh ($14) relative to the continuous system. With less space conditioning energy, the distribution energy use was even more dominant for the HRV and continuous supply than in Houston and made them the least energy efficient. The higher ventilation rates of the continuous supply and continuous exhaust plus intermittent supply lead to greater space conditioning energy use. Although the HRV has the greatest air change rates, the recovery of energy is shown by having the least space conditioning energy use. However the distribution energy requirements and ventilation fan power make the HRV the greatest energy user: requiring an additional 2250 kWh compared to the standard unventilated house (costing an additional $240 compared to the standard house). As with Houston, the leaky house used more energy than either of the exhaust systems.

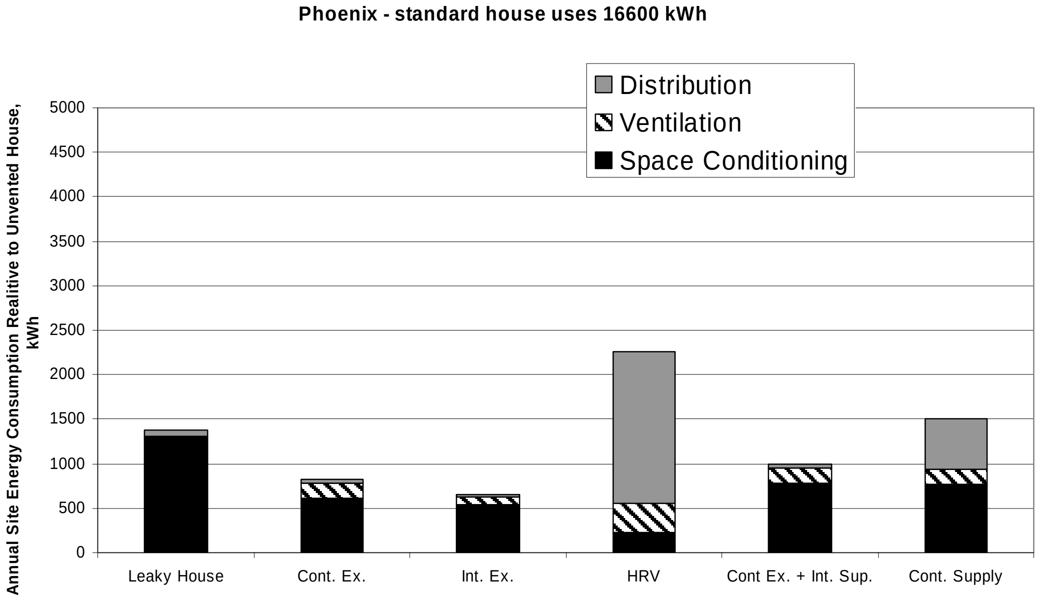

Figure 2. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Phoenix.

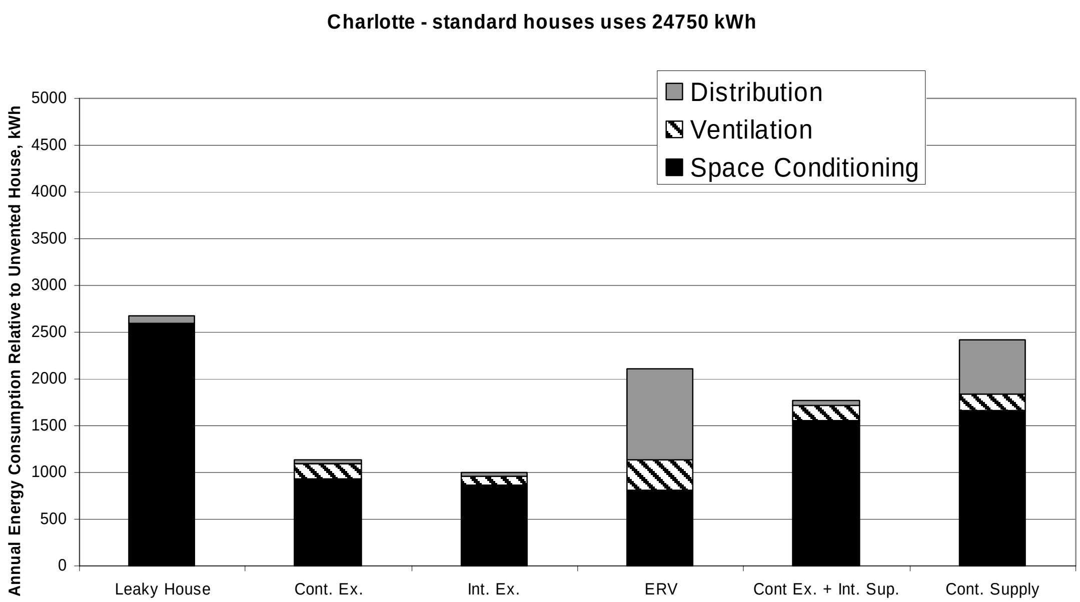

Figure 3. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Charlotte.

Figure 3 shows the results for Charlotte where the total energy use for the standard house was 24,750 kWh (cost was $1410). For Charlotte, the less benign climate led to the energy for space conditioning increasing significantly - in particular for the higher ventilation rate systems and the leaky house. The lowest energy cost exhaust system used 1150 kWh ($75) more than the unventilated house. The intermittent exhaust saved only 45 kWh (less than $10) relative to the continuous system. The leaky house was the greatest energy user, followed by the continuous supply. The ERV had the lowest space conditioning energy use that made it more energy and cost competitive with the other systems.

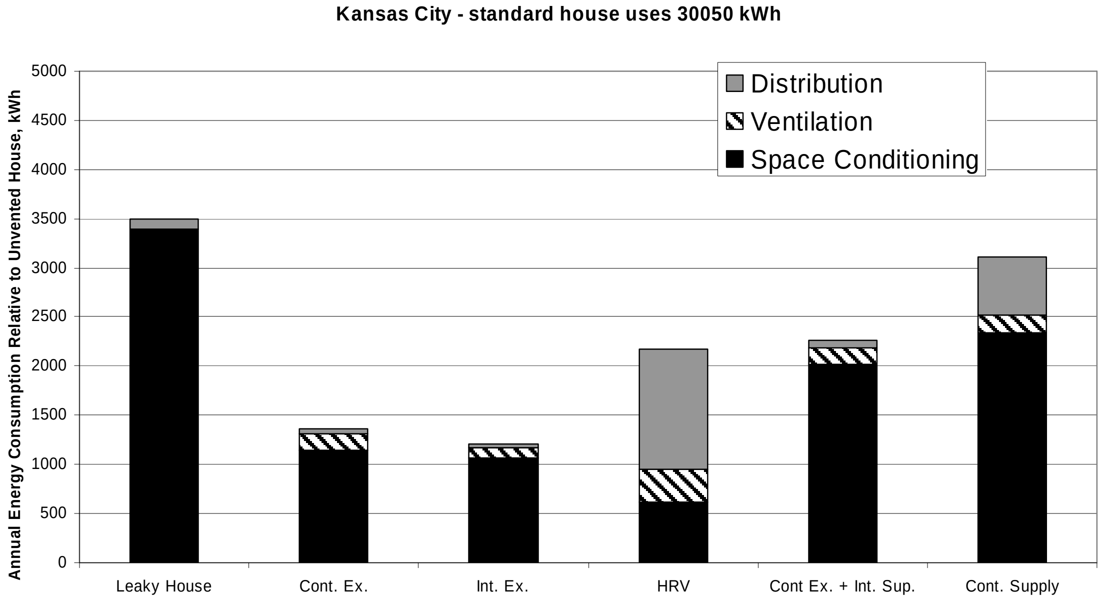

Figure 4. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Kansas City.

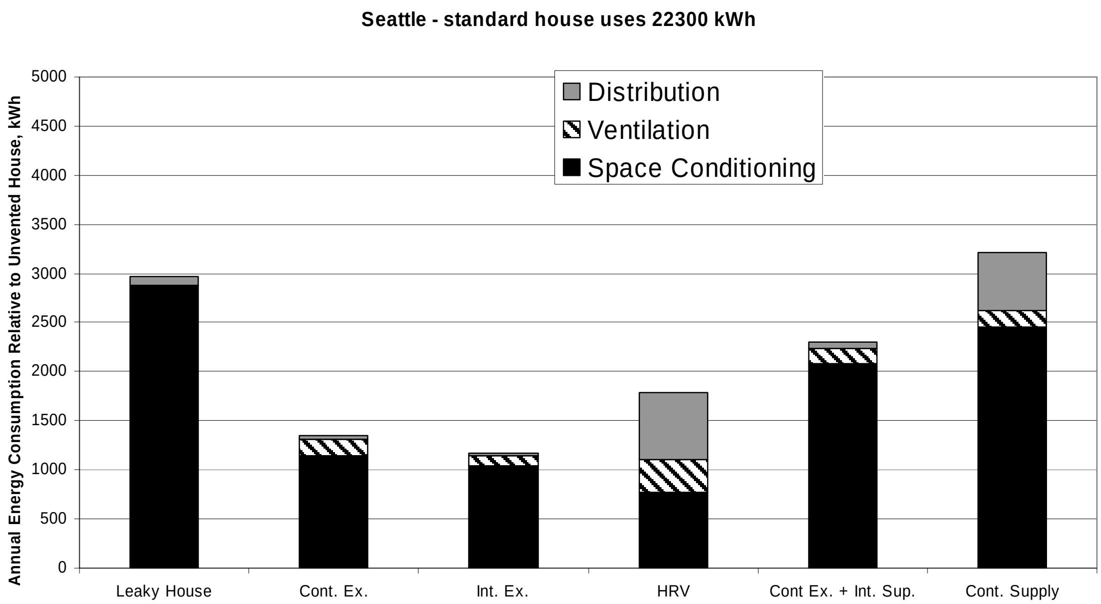

Figure 5. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Seattle.

For Kansas City, the climate is heating-dominated, so cooling energy use is only about 5% of the total. The colder climate results in the space conditioning energy use that is higher relative to the energy for distribution and ventilation. The same trends are seen as for Charlotte, with the leaky house requiring the most additional energy (3500 kWh ($190)) and the continuous exhaust system the least (1365 kWh ($85)).

Seattle uses very little air conditioning - compressor energy was less than 1% of the gas energy used. However, the milder winter climate compared to Kansas City led to less total energy use and less difference between the unventilated standard house and the ventilation systems. The distribution energy for the HRV and Continuous exhaust with intermittent supply are lower due to the small capacity furnace and air conditioner required in Seattle and its smaller blower.

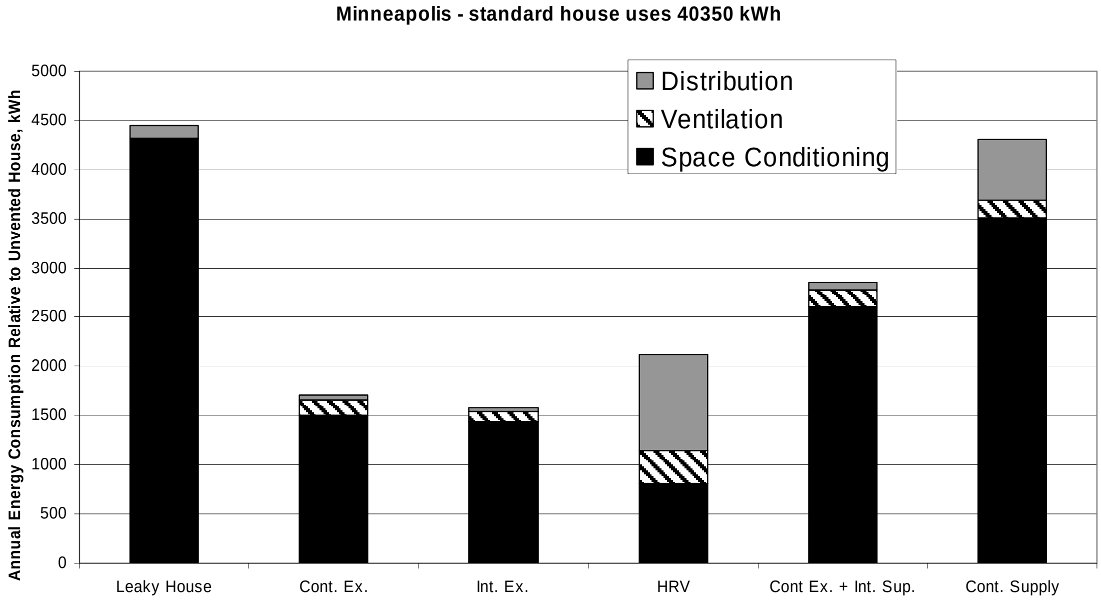

Figure 6. Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Minneapolis.

Minneapolis was the harshest climate investigated in this study and used the most energy. The low outdoor temperatures in winter make the leaky house a big energy user - requiring 4400 kWh ($240) more than the standard house. Although the space conditioning energy use for the HRV is about half that for the exhaust systems, the blower energy used to distribute the air is sufficient that it requires 400 kWh ($60) more energy. The higher air change rates of the continuous exhaust plus intermittent supply and the continuous supply lead to much more energy use than the exhaust only cases in this extreme climate.

Exhaust systems were found to be more energy efficient for several reasons: they had the lowest ventilation rates, they used the least ventilation fan power and they did not use fans to mix or distribute air. Since the exhaust system had a lower impact on the total air exchange, its space conditioning impact was smaller and it used the least energy (and had the least operating cost) by virtue of producing fewer air changes. Although the exhaust systems had the lowest average ventilation rates the continuous exhaust still provided the required minimum ventilation rate when natural infiltration was zero (or close to zero). As natural infiltration increases, the added infiltration for imbalanced systems decreases. This is a good strategy for minimizing energy impacts while maintaining a suitable minimum air flow.

Air distribution systems are used to distribute heating and cooling, to distribute fresh air, and to filter, clean and recirculate indoor air. As such, they can be seen as a separate building function from the rest of the ventilation system and could be treated separately both in regulation and design. One advantage of treating distribution as a separate function is that it will allow efficiency advances in distribution to be made independently from, for example, cooling systems. Other studies (Pigg and Talerico (2004) and Gusdorf et al. (2002)) have shown that using more efficient variable speed motors at low speed to circulate and mix air in a house results in significant energy savings relative to the blowers in most systems like the ones used in this study. Other options include using the HRV/ERV fans to also distribute the air in the house either through a dedicated duct systems or using the existing forced air system ducts.

HRV/ERV systems can also be configured to use the forced air heating and cooling distribution ducts but without using the central furnace or air conditioning blower. This has the potential to reduce the energy used to distribute the air (although this must be balanced in heating by the reduction of electric motor heat (several hundred watts)). In the results shown in Figures 1 through 6 the reduction in distribution energy means than an HRV/ERV uses about the same energy as the exhaust systems in the moderate climates and may actually use less energy in the colder climates. However, there is potential for short-circuiting of HRV/ERV supply air through the central furnace/blower to the ERV/HRV return ducting thus bypassing the house and reducing the effective ventilation rate that would need to be accounted for in such a comparison. This short circuiting issue is a suitable topic for future study.

CONCLUSION AND RECOMMENDATIONS

In this paper we used a simulation approach to determine the likely energy impacts of specific residential ventilation requirements and strategies for six different climates. The energy to provide such minimally acceptable ventilation was in the range of 800 to 1700 kWh per year ($50 to $100) for the most efficient options, which typically represented about 5% of the heating and cooling energy.

The space conditioning energy associated with the extra ventilation provided by the ASHRAE 62.2 ventilation systems dominated their energy use, except for systems that distributed the air.

In every climate the intermittent exhaust system proved to be the most energy efficient system for meeting the proposed requirements—followed by the continuous exhaust system. The simplest and most cost-effective method to comply with ASHRAE 62.2 was to use continuous exhaust fans.

Existing homes with leaky envelopes paid a significant energy penalty without necessarily providing more ventilation then a house with a better envelope and mechanical ventilation.

Exhaust systems were found to be more energy efficient than the alternatives because the requirements of 62.2 do not correctly account for the addition of infiltration and mechanical ventilation. Since the exhaust system had a lower impact on the total air exchange, its space conditioning impact was smaller and it was the more cost-effective approach by virtue of producing fewer air changes.

Systems that used the furnace or air conditioner blower to distribute air used substantially more energy than other systems. In particular for HRVs and ERVs, the use of the central blower counteracted the space conditioning energy savings of these devices such that they used more energy than a simple exhaust system. HRV/ERVs can be installed in configurations that do not use the central blower, in which case their energy use will be similar to exhaust only systems and may be even less in cold climates.

Although the ventilation related energy required changed by about a factor of three between mild and harsh climates, the general trends for relative energy use between systems was consistent.

The results of this study indicated that the following research needs exist:

- A method to allow for the different contributions of exhaust, supply and balanced systems should be considered for use in ASHRAE 62.2.
- A broad based field study to determine envelope and duct air leakage of current new construction and how commonly used mechanical ventilation systems perform with such leakage.
- Evaluation of possible ventilation credit to be given to ventilation systems that distribute air or develop a separate efficiency requirement for air distribution.
- Evaluation of short circuiting potential for HRV/ERVs using heating/cooling ducts that do not have simultaneous central blower operation.

ACKNOWLEDGEMENTS

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Building Technology, State and Community Programs, Office of Building Research and Standards, of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098. This work was also supported in part by the Air-conditioning and Refrigeration Technology Institute under ARTI Research project no. 614-30090. The authors would like to thank Armin Rudd, Steven Emmerich, Roy Crawford, Wayne Reedy and Steve Syzmurski for their contributions to this study.

REFERENCES

ACCA Manual J - Load Calculation for Residential Winter and Summer Air Conditioning. Air Conditioning Contractors of America, Washington, DC.

ACCA Manual S - Residential Equipment Selection. Air Conditioning Contractors of America, Washington, DC.

ALA. 1999. Indoor Air Pollution Fact Sheet. American Lung Association.

ASHRAE Handbook of Fundamentals. 2005. ASHRAE, Atlanta, GA.

ASHRAE Standard 62.2, 2004, "Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings," American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

ASHRAE Standard 136. "A Method of Determining Air Change Rates in Detached Dwellings. American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

ASHRAE Standard 152, 2004, “Method of Test for determining the design and seasonal efficiencies of residential thermal distribution systems” American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

ASHRAE Standard 160P (Draft). 2006. *Design Criteria for Moisture Control in Buildings* American Society of Heating, Refrigerating and Air conditioning Engineers, Atlanta, GA.

Building Science Corporation. 2006. *Analysis of Indoor Environmental Data*. Building Science Corporation, Westford, MA.

EPA. 2001. *Indoor Humidity Assessment Tool (IHAT) Reference Manual*. Environmental Protection Agency, Washington, DC.

Gusdorf, J., Swinton, M., Entchev, E., Simpson, C., and Castellan, B. 2002. The Impact of ECM furnace motors on natural gas use and overall energy use during the heating season of the CCHT research facility. National Research Council Canada report No. NRCC-38443.

Henderson, H.I. and Rengarahan, K. 1996. A model to Predict the Latent Capacity of Air Conditioners and Heat Pump at Part-Load Conditions with Constant fan Operation. ASHRAE Trans, Vol. 102, Pt. 1, pp. 266-274. ASHRAE, Atlanta, GA.

Henderson, H.I. 1998. The Impact of Part-Load Air-Conditioner Operation on Dehumidification Performance: Validating a Latent Capacity Degradation Model. Proc. IAQ and Energy 1998. pp. 115-122.

HVI. 2005. Certified Home Ventilating Products Directory, Home Ventilating Institute. Wauconda, IL.

ICC. 2005. “International Energy Conservation Code.” International Code Council, Country Club Hills, IL.

McWilliams, J., Sherman M.. 2005. “Review of Literature Related to Residential Ventilation Requirements”. LBNL 57236. Lawrence Berkeley National Laboratory, Berkeley, CA.

Palmiter, L. and Bond, T., (1991), “Interaction of Mechanical Systems and Natural Infiltration”, Proc. 12th AIVC Conference, Ottawa, Canada, September 1991. pp. 285-295.

Pigg, S. and Talerico, T. 2004. Electricity Savings from Variable-Speed Furnaces in Cold Climates. Proc. ACEEE Summer Study 2004, pp. 1-264 – 1.278. American Council for an Energy Efficient Economy, Washington, DC.

Price, P.N. and M.H. Sherman "Ventilation Behavior and Household Characteristics in New California Houses," LBNL-59620. Lawrence Berkeley National Laboratory. Berkeley, CA.

Rudd A and Henderson, H., 2006. “Monitored Indoor Moisture and Temperature Conditions in Humid US Climates,” To be published ASHRAE Trans.

Russell, M. Sherman, M.H. and Rudd, A. 2005. “Review of Residential Ventilation Technologies”, LBNL 57730. Lawrence Berkeley National Laboratory, Berkeley, CA.

Sherman, M. H. 2004. “Efficacy of Intermittent Ventilation for Providing Acceptable Indoor Air Quality” (In review at *HVAC&R Research Journal*). LBNL 56292. Lawrence Berkeley National Laboratory, Berkeley, CA.

Sherman, M.H. and Matson, N.E., 2002. “Air Tightness of New U.S. Houses: A Preliminary Report”, Lawrence Berkeley National Laboratory, LBNL 48671. Lawrence Berkeley National Laboratory, Berkeley, CA.

Sherman, M. and D. Dickerhoff,1994 "Air-Tightness of U.S. Dwellings'' In Proceedings, 15th AIVC Conference: The Role of Ventilation, Vol. 1, Coventry, Great Britain:Air Infiltration and Ventilation Centre, pp. 225-234. (LBNL-35700)

Sherman, M.H. and D.J. Wilson, "Relating Actual and Effective Ventilation in Determining Indoor Air Quality," Building and Environment 21 (3/4): 135-144, 1986.

Walker, I.S., Wilson, D.J., and Forest, T.W., (1996), “A Wind Shadow Model for Air Infiltration Sheltering by Upwind Obstacles”, ASHRAE HVAC&R Research Journal, Vol. 2, No. 4, pp. 265-283, ASHRAE, Atlanta, GA.

Walker, I.S. and Sherman, M.H. 2006. “Evaluation of Existing Technologies for Meeting Residential Ventilation Requirements”, LBNL 59998. Lawrence Berkeley National Laboratory, Berkeley, CA.

Walker, I.S., Forest, T.W. and Wilson, D.J. 2005. “An Attic-Interior Infiltration and Interzone Transport Model of a House”, Building and Environment, Vol. 40, Issue 5, pp. 701-718, Elsevier Science Ltd., Pergamon Press, U.K.

Walker and Sherman, M.H. 2007. “Humidity Implications for Meeting Residential Ventilation Requirements”, Proc. DOE/BTECC/ASHRAE Conference on the Thermal Performance of the Exterior Envelopes of Biuldings X, LBNL 62182.

Wilson, D.J., and Walker, I.S., (1990), "Combining Air Infiltration and Exhaust Ventilation", Proc. Indoor Air '90, July 1990, Toronto, Canada. pp.467-472.

Walker, I.S. 1998. Technical Background for Default Values used for Forced Air Systems in Proposed ASHRAE standard 152P. *ASHRAE Trans. Vol.104 Part 1*. LBNL 40588.

Yuill, G.K. “The variation of the effective natural ventilation rate with weather conditions”, *Renewable Energy Conference ’86*. Solar Energy Society of Canada Inc. pp. 70-75, 1986.

Yuill, G.K. “ The development of a method of determining air change rates in detached dwellings for assessing indoor air quality. *ASHRAE Transactions* 97(2). 1991.